APPLICATION OF THE AIR-CONDITIONING SYSTEM ENERGY SIMULATION FOR COMMISSIONING (ACSES/CX) TOOL TO HVAC SYSTEM COMMISSIONING

Part 2: Application to the Substation of a Heat Source System with Bleed-in Control

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ABSTRACT

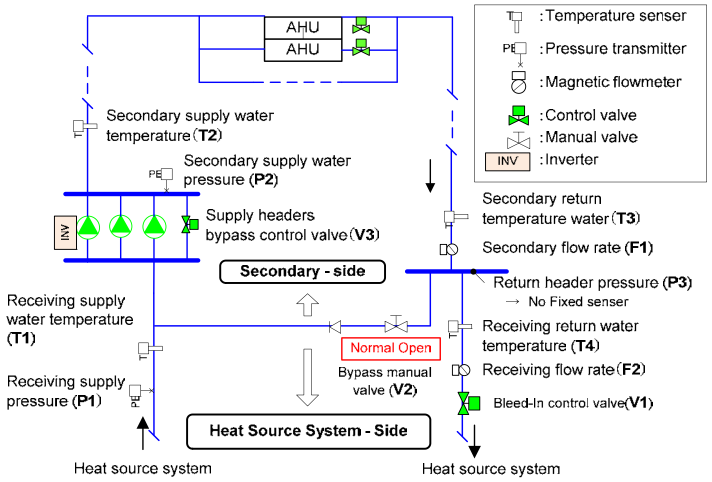
Simulation tools are effective for the commissioning of HVAC systems. In particular, such tools are useful not only for simulating the current state of systems but also for simulating the performance of a system provided with efficiency improvements. The Air-Conditioning System Energy Simulation for Commissioning (ACSES/Cx) tool is a software tool developed to provide specialized functions for the commissioning process. In the present paper, the ACSES/Cx simulation tool is applied to the substation of a secondary heat source system and secondary air-handling units (AHUs) of a large-scale building. We attempt to show how the ACSES/Cx tool can be used to simulate actual sophisticated systems with sufficient accuracy with a simplified model and can be used to evaluate energy-saving improvements or renovation of the system.

INTRODUCTION

A very small temperature difference between the supply and return chilled water often occurs in HVAC systems. As such, many systems supply excess chilled water, resulting in wasted energy. Common causes of this problem include inappropriate set-points of pressure control for supply pumps, excessively low temperature set-points beyond the capacity of the AHU, and performance deterioration of the AHU cooling coil. However, in the present study, we would like to investigate an additional problem caused by bleed-in (BI) control. BI control is commonly equipped in the substations of district heating and cooling (DHC) plants in order to maintain the designed temperature difference between the supply and return chilled water in order to enable efficient operation of the main power plant. In order to investigate this problem, we constructed a simulation model of the secondary air-conditioning system, including the cooling coils of AHUs. Although the system consists of approximately 250 AHUs, we found that simplification is possible by categorizing AHUs into six groups based on measurement data analysis, and effective commissioning can be performed.

COMMISSIONING PROJECT

The commissioning project considered herein is a large building complex in Kyoto, Japan, which has 16 floors above ground and three basement floors



Unit	Pump specifications	Number of pumps
Secondary pumps	540 m^3/h × 35 m × 75 kW	3 (One of which has an inverter)

* Measurement points such as T1, T2, …, P1, …, are indicated by bold letters. These abbreviations are used hereinafter.

Figure-1 Secondary chilled water pump system for upper floors in the department store substation

Table-1 Control strategy and set-points

Control item	Control outline	Set-point
Number control of secondary pump	Control of increase and decrease the number of pump depending on secondary flow rate (F1)	Increase from one to two : 627 m^3/h
		Increase from two to three : 1188 m^3/h
Secondary supply pressure constant control	PID control of pump inverter and bypass valve between supply headers (V3) depending on secondary supply pressure (P2). But the valve (V3) is shut completely until inverter becomes the Lower limit level.	Pressure set-point : 0.86 MPa
		Lower limit of inverter output (With only one pump operating): 70 %
		Lower limit of inverter output (With more than two pumps operating): 90 %
Bleed-in control	PI control of bleed-in control valve (V1) depending on receiving return temperature (T4)	Temperature set-point : 12.8 ℃
		Minimum opening position of V1 : 16 %

with a total floor area of 235,942 m^2. The total refrigerator capacity of this building is 26.3MW, and chilled water is supplied to six substations, which are located in remote areas far from the central heat source plant, such as department stores, hotels, theatres, and train stations. A substation located in a department store was chosen as the study target for the commissioning because the cooling load demand throughout the year is the highest among all substations. This substation has two sub-systems for upper floors and lower floors. We herein analyze only the upper-floor sub-system. Figure 1 shows the secondary chilled water pump system of the substation for the upper floors. The automatic control strategy and the control set-points are shown in Table 1. The pressure of the chilled water is controlled by controlling the speed of an inverter pump and the opening of a bypass control valve (V3) located between the primary and secondary supply headers.

When only one inverter pump is operating, the minimum inverter set-point giving speed is 70% (42 Hz), which may not be low enough. As described in the introduction, BI control is a temperature compensation control method for maintaining the receiving return water temperature (T4). The return water control valve, BI valve (V1), is controlled by proportional-integral-derivative (PID) control in order to make the set-point, 12.8°C. The minimum opening position of V1 is set to 16%.

PERFORMANCE VERIFICATION

We verified the performance of the secondary pump system of the substation using data that was recorded hourly from January to December, 2010, taken from the building energy management system (BEMS).

Frequency Analysis of Secondary Flow Rate

The occurrence frequency of secondary flow rate (F1) throughout the year is shown every 20% in Figure 2. The rated flow rate for one pump (540 m^3/h) is assumed to be 100%. The number of occurrences over the course of a year is shown in the figure. The occurrence frequency under the rated flow rate for one pump reaches 60%. For this reason, the operation time for the low-flow-rate range is very long.

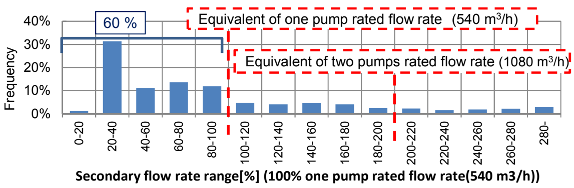


Figure-2 Frequency of secondary flow rate

Water Temperature Analysis

The receiving supply temperature (T1) of the secondary flow rate (F1) range, the secondary supply water temperature (T2), and the secondary water temperature difference (T3-T2) are shown in Figure 3. Here, T2 became higher than T1 over the entire flow rate range by closing the BI control valve (V1) in order to maintain the receiving return temperature (T4). As a result of closing V1, a large amount of secondary return water flowed into the bypass pipe between the supply and return water. This generates an increase in T2 because the bypass water from the return side and the receiving supply water are mixed. The temperature increase of the secondary supply water caused an efficiency degradation of AHU coils and, consequently, an additional increase in the demand flow rate. The temperature difference (T3-T2) was relatively small for the entire flow rate range and became less than 3 K for operation below the rated flow rate for one pump (less than 100% of the flow rate range).

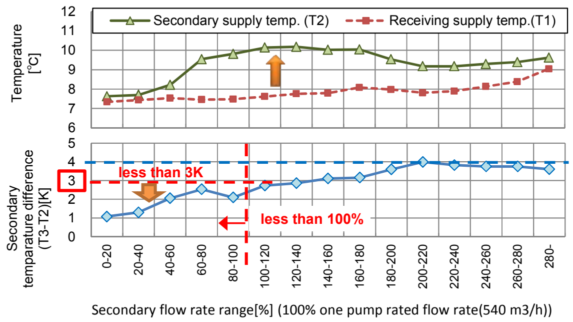


Figure-3 Temperatures T2 and T1 and the temperature difference (T3-T2) at various points in secondary flow rate range

Water Pressure Analysis

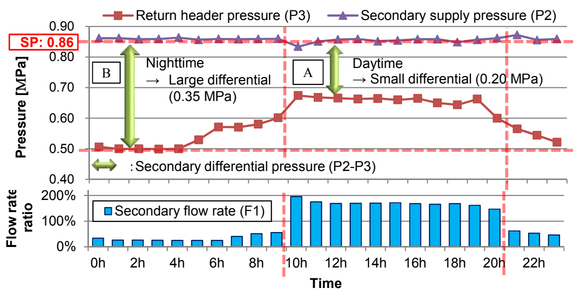


Figure-4 Hourly mean values of secondary flow rate and pressure in July

The load increases particularly in July. Figure 4 shows the data for hourly averages in July for the secondary supply pressure (P2), the return header water pressure (P3), and secondary flow rate (F1).

Having direct effects to the secondary water circulation is, in fact, not P2 but secondary differential pressure (P2-P3, shown as a double-headed arrow in the figure). P2 is controlled in any hour by 0.86 MPa of set-point, but P3 increases significantly in the daytime when the secondary flow rate rises. Therefore if the secondary differential pressure (P2-P3) at daytime is appropriate, the differential pressure at night is higher than that during the day and is excessive, despite the flow rate is low. As the flow rate is low during seasons other than summer and night, extremely high differential pressure is thought to be supplied to the secondary side in the same situation described above with the exception of daytime in the summer.

Performance Analysis of Air-handling Units

The AHUs include 21 fresh-air-handling units (FAHUs) and 234 room air circulation type terminal air-handling units (ATUs). The total supply air volume of the FAHUs is 378,480 m^3/h, and the ATU total supply air volume is 1,270,800 m^3/h. The water quantity of each unit is controlled by a two-way valve. During midsummer, for all FAHUs and ATUs, we analyzed data related to set-points (FAHU: supply air, ATU: return air) and data related to the opening positions of valves controlled by the set-points of all FAHUs and ATUs. Some FAHUs and ATUs were operated by an abnormal air temperature set-point such as 10°C FAHU supply air (design temperature: 16°C) and 20°C ATU return air (design temperature: 26°C). As a result, the opening positions of the chilled water valves were continuously fully open (100 %) and out-of-control, as shown in Figure 5. Counting by cooling capacity ratio 24% of FAHUs and 66% of ATUs were uncontrollable. These FAHUs and ATUs were set to abnormally low temperatures because the temperature increase of the secondary supply water described in the water temperature analysis section and performance deterioration of the coil due to coil fouling.

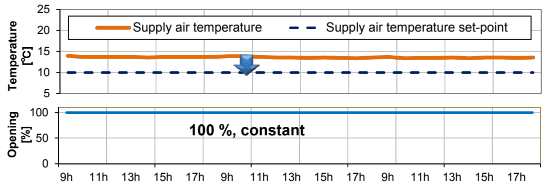


Figure-5 Situation of an uncontrollable FAHU

Summary of Problems

Figure 6 shows a flowchart summarizing the problems that were detected by the present analysis. This figure shows how the problems occurred during the operation stage.

[Cause of Problem]

- Substation side:
  - Excessive supply pressure of chilled water
- AHU side:
  - Inappropriate set-point of air temperature beyond the cooling coil performance
  - Performance deterioration of the cooling coil

[Present Issues]

- Secondary flow rate is excessive
- Temperature increase of secondary supply water
- Decrease in secondary temperature difference
- Energy loss in secondary pumps

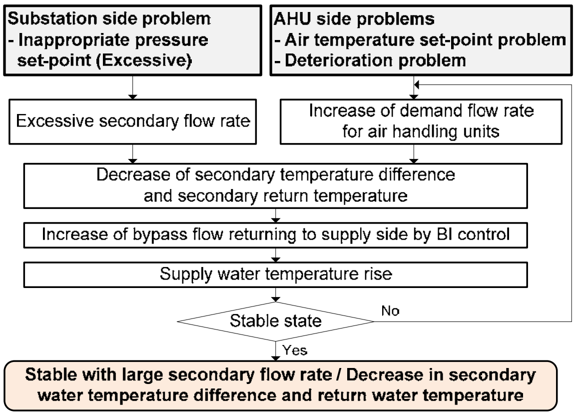


Figure-6 Problem occurrence flowchart

SIMULATION MODEL BY ACSES/CX

Model of the Entire Secondary Air Conditioning System

Simulation tool called ACSES/Cx was used to build a model of the entire secondary air conditioning system. The outline of ACSES/Cx and the characteristics are described in our part-1 paper.

Based on a performance diagram provided by manufacturers, we firstly need to input parameters of equipment models, such as a cooling coil model and a pump model, as well as control logic models, such as a pump number control model and a BI control model. In ACSES/Cx these models are connected to form the network of the entire secondary air conditioning system. The model of the entire secondary air conditioning system is comprised of the following three subsystem models:

- AHU model
- Bleed-in control model
- Chilled water pump system model

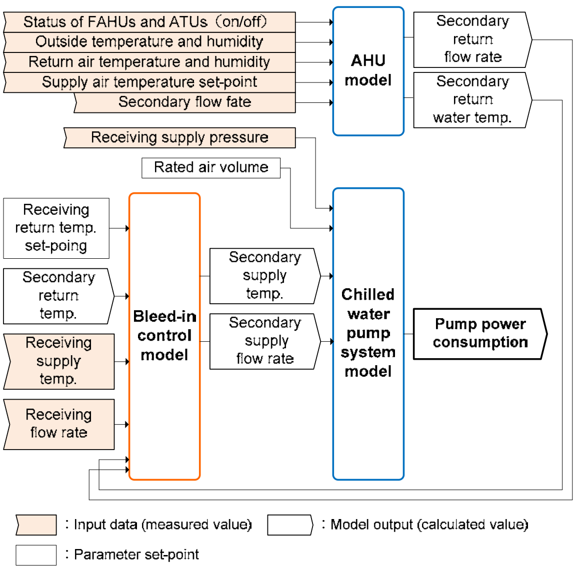


Figure-7 Model of the entire secondary air conditioning system

Figure 7 shows the model of the entire secondary air conditioning system. The inputs of the model of the entire secondary air conditioning system are external weather condition data, the cooling heat load, the receiving flow rate (F2), the receiving supply water pressure (P1), the signal status of FAHUs and ATUs, the return air temperature and humidity of FAHUs and ATUs, and the air temperature set-point of FAHUs and ATUs. The final output point is the chilled water pump power consumption. The BEMS data have all input points, and the data are actual measurement data. The calculation time interval for simulation is two minutes. The three subsystem models mentioned above are described in the following.

Component Models

- **Air-handling Unit**

The secondary air-conditioning system is a sophisticated system formed by numerous AHUs, e.g., 21 air-handling units for fresh air treatment (FAHU) or 234 air-handling terminal units for room air temperature control (ATU). Table 2 shows classification results of the AHUs based on whether they were operated in good control or out-of-control. The ratio of out-of-control AHUs, which control valves are always fully open, and AHUs operated in good control is also shown, where the ratios are the percentage of the heat capacity of the classified AHUs. As there are two types of fresh-air-handling

Table-2 Grouping of AHUs

Air handling unit (AHU) type		Ratio of capacity	Exsisting control condition	Ratio of capacity
Fresh air handling unit (FAHU)	With total heat exchanger	13%	Good contorl	10%
			Ouf of contorl	4%
	With no total heat exchanger	33%	Good contorl	26%
			Ouf of contorl	7%
Terminal air handling unit (ATU)		54%	Good contorl	17%
			Ouf of contorl	37%

units, that is, with a total heat exchanger (HEX) and without the HEX, classification is provided for each one. As a result more than 250 AHUs are classified into six types.

Figure 8 shows how the categorized AHUs models are connected to process the given heat load. The processing methods for this integrated model are described below.

1) Input the input data (Figure 8(*1)) into four FAHU models and calculate the load processed by the FAHUs (Figure 8(*2)) and the temperature and flow rate of outlet chilled water.

2) Input this calculated load (*2) into the ATU load calculating model which calculates ATU heat load by subtracting the input load (*2) from the given total heat load (*3).

3) Input this ATU heat load into the out-of-control ATU model which temperature set-point is too low to control the chilled water flow rate. This means the flow rate is out-of-control or maximal.

4) Input the remainder of ATU load that could not be processed by the out-of-control ATU model into the good control ATU model and perform load processing.

5) Input these calculation results for flow rate (Figure 8(*4)) and outlet temperature (Figure 8(*5)) from each of the four FAHU models and two ATU models into the total AHU load calculation model, in which the total flow rate of return chilled water is calculated by adding the values of six flow rates, and the temperature is calculated by taking the weighted average of the six outlet temperatures for each flow rate.

We used the SIMBAD detailed static cooling coil, which has been proven to be a high-accuracy model, as a cooling coil model of the AHU. This cooling coil

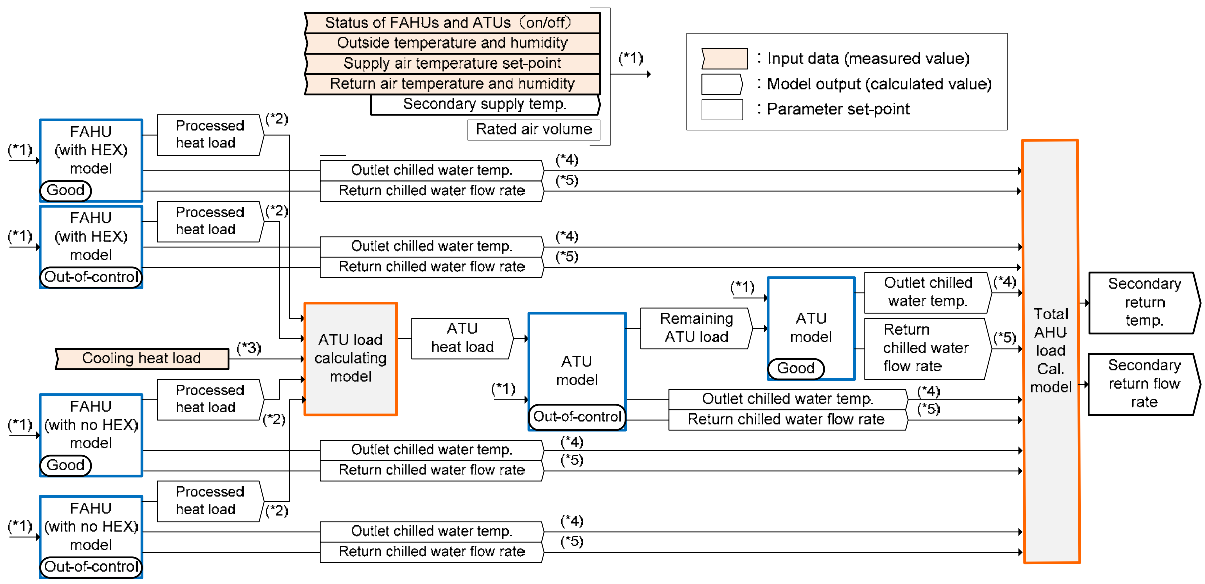


Figure-8 AHU model

model considers three cases: dry, wet, and wet-and-dry. The quantity of total exchange heat (Qt) and the bypass factor (Bf) are calculated as input values based on the inlet status of water and air. Furthermore, each outlet status is calculated using Qt and Bf and the heat balance equation of the air side and the water side, as follows:

$$Q_t = \varepsilon_c \cdot C_{\min(\theta_{ai}-\theta_{wi})} \tag{1}$$

$$B_f = \exp\left(\frac{-A_{ext}}{C_{air}} \cdot R_a\right) \tag{2}$$

$$\varepsilon_c = f(U_a, C_{air}, C_{water}) \tag{3}$$

$$U_a = \frac{A_{ext}}{R_a + R_m + R_w} \tag{4}$$

$$C_{\min} = \min(C_{air}, C_{water}) \tag{5}$$

where ε_c is the coil efficiency [-], A_{ext} is the front area of the coil [m^2], U_a is the total heat transfer coefficient [$kW/m^2 \cdot K$], R_a is the thermal resistance of air [$m^2 \cdot K/kW$], R_m is the thermal resistance of metal [$m^2 \cdot K/kW$], and R_w is the thermal resistance of water [$m^2 \cdot K/kW$].

- **BI Control**

Figure 9 shows the BI control model flowchart. The processing method is as follows:

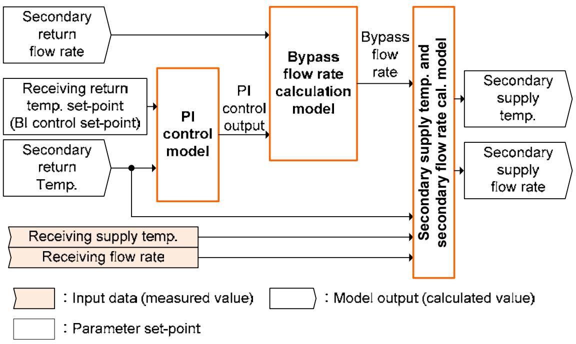


Figure-9 Bleed-in control model

1) Input the secondary return temperature (T3) into the PI control model of the BI control valve (V1) and output its controlling value of V1. The bypass flow rate is controlled by PI to reach the set-point temperature of receiving return water (T4).

2) Input the controlling value of V1, which is the output of 1) into the bypass flow rate calculation model, and output the bypass flow rate. The secondary return flow rate (F1) can be distributed to the bypass flow rate and the receiving flow rate (F2) by calculating the bypass flow rate using this model.

3) Input the bypass flow rate, the secondary return temperature (T3), which is temperature of the bypass flow, the receiving supply temperature (T1), and the receiving flow rate (F2) into the secondary supply temperature and flow rate calculation model. Output the secondary supply temperature (T2) and the secondary supply flow rate (secondary flow rate (F1)). These output values become the input values of the AHU model.

- **Chilled Water Pump System**

Figure 10 shows a flowchart of the chilled water pump system model. The processing method is as follows:

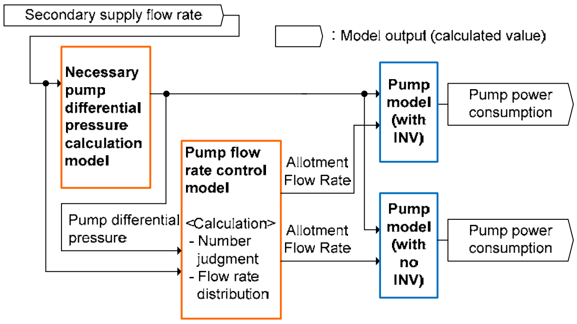


Figure-10 Chilled water pump system model

1) Input the flow rate of the secondary supply water into the differential pressure calculation model of the necessary pumps. Calculate the required pump differential pressure and the secondary water differential pressure needed to circulate the flow rate appropriately.

2) Input the secondary supply flow rate and the pump differential pressure into the pump flow rate control model. Distribute the flow rate to the pump after determining how many operation units are required. If only one pump is required, all flow rates are input to the pump with the inverter. On the other hand, if more than two pumps are required, the flow rate is distributed among the pump with the inverter and the pumps without inverters.

3) Calculate and output the individual power consumptions of the pump with the inverter and of the pumps without inverters by the pump flow rate control model and the pump model without inverter.

Regarding the calculation of required pump differential pressure to the flow rate of 1), we use the relational expression between the flow rate and the required pump differential pressure obtained from the experiment carried out using the real AHU system. In this experiment, various conditions were set under the situation that the supply-return header bypass manual valve was closed completely and the BI control valve was forcibly fully open. The differential pressure at the control valve of the most far FAHU from the substation became 40 kPa (which is required for flowing with an appropriate amount of water through the FAHU) by adjusting the number of pumps and inverters at each flow rate point. The measurement value of secondary differential pressure at that time was supposed to be the required pump differential pressure. The results of this experiment are shown in Figure 11 in conjunction with measurement data for 2010.

Comparing the results revealed that the secondary differential pressure, i.e., the pump differential pressure, was excessive. The pump differential pressure input into the pump model is given by the quadric approximation formula of the experiment data, as indicated by the approximation formula shown in Figure 11. The following formula, which is adopted in HVACSIM+, is used for the model formula of the pump:

$$C_k = k_{ch}(a_{ch,0} + a_{ch,1}C_f + a_{ch,2}C^2{}_f + a_{ch,3}C^3{}_f + a_{ch,4}C^4{}_f) \tag{6}$$

$$\mu_f = k_\eta(a_{\eta,0} + a_{\eta,1}C_f + a_{\eta,2}C^2{}_f + a_{\eta,3}C^3{}_f + a_{\eta,4}C^4{}_f) \tag{7}$$

where C_k is the non-dimensional differential pressure [-], η_f is the equipment efficiency [-], and C_f is the non-dimensional flow rate [-].

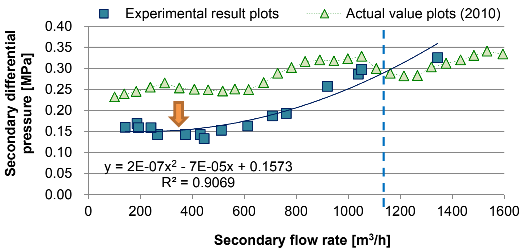


Figure-11 Plots of actual values for 2010 and experimental results for secondary differential pressure with respect to secondary flow rate

ACCURACY OF THE SIMULATION MODEL

Accuracy of the Component Model

We investigated the simulation accuracy of the pump system model and the cooling coil model used in the entire secondary air conditioning system model. Figure 12 compares the measurement and simulation values of total pump power consumption from June to September. The simulation error is very small, approximately 3%.

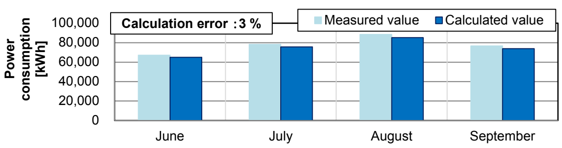


Figure-12 Measured and simulated pump power consumptions

In order to verify a cooling coil model, Figure 13 shows the variation with time for actual measurement data and simulated values of supply air temperature and cooling coil outlet water temperature for two days. The mean square error is 0.16°C for the supply air temperature and 0.40°C for the coil outlet water temperature. The simulation accuracy is sufficient in order to use these models in simulation.

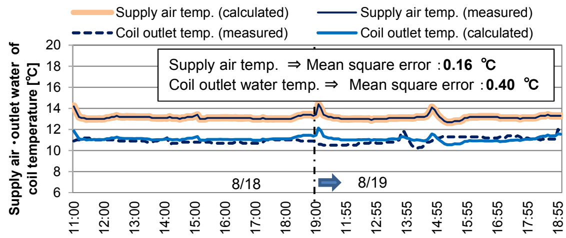


Figure-13 Measured and simulated supply air temperature and coil outlet water temperature

Accuracy of the Entire Secondary Air Conditioning System Model

The accuracy of the model of the entire secondary air conditioning system was verified for three days, from August 17th to 19th, as an example. Variations with time for the measured and simulated vales are shown along with the average errors and the mean square errors in Figure 14 for pump power consumption and in Figure 15 for the secondary water flow rate. The simulation accuracy is sufficient for the present investigation.

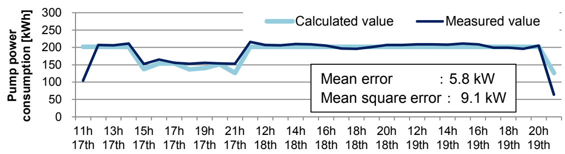


Figure-14 Measured and simulated pump power consumption

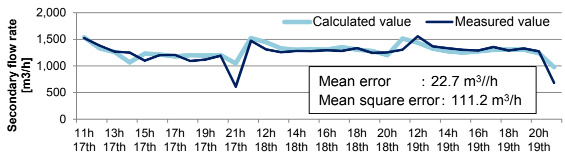


Figure-15 Measured and simulated secondary flow rate

IMPROVEMENT BY SIMULATION

Improvements

We implemented five improvement plans, as described below, in order to reduce the chilled water pump power consumption. The improvements were drafted in order to improve the shortcomings revealed by the performance investigation. Improvements to the substation, including the secondary pump system and BI control, are presented below as A-series (items 1 through 3), and improvements to the AHU system are presented below as B-series (items 4 and 5). Only item 5 is an improvement to heat load reduction, and the other heat loads are the same as the current load. Using

ACSES/Cx, we clarify the effects of these improvements through simulation by modeling the entire secondary air-conditioning system, including the substation.

1) Improvement of Pressure Control Method
(Improvement A-1)

The set-point of the pump inverter is currently controlled as a fixed pressure of the secondary supply water (P2). We change this point to a variable set-point, which is given by an appropriate secondary differential pressure (P2-P3), depending on the demand secondary flow rate (F1). Regarding the demand flow rate and secondary differential pressure, we used an experimentally approximate expression as a relational expression. The minimum inverter output is reduced from 70% (42 Hz) to 20% (12 Hz) when only one pump is operated.

2) Removing BI control
(Improvement A-2)

BI control is removed. In other words, the bypass manual valve between the return header and the supply side (V2) is completely closed, and the BI control valve (V1) is always open.

3) Inverter installation for all secondary pumps
(Improvement A-3)

When more than two pumps operate, the inverter output cannot be reduced. Therefore inverters are installed in all secondary pumps in order to improve this defect.

4) Adjustment of AHU Control
(Improvement B-1)

A number of the AHUs (FAHUs, ATUs) are out-of-control. Therefore, we adjust the air temperature set-point of all of the AHUs to their designed set-point and reduce deterioration of air conditioners in order to realize their design performance.

5) Adjustment of Volume of Fresh Air
(Improvement B-1)

The measurement results for CO_2 concentration in the room reveal that the volume of fresh air can be reduced by 30%. Therefore, we reduce the fresh air volume by 30%.

Simulation Cases

Using the simulation model, we examined the reduction effects of secondary pump power consumption due to five improvements described in the previous section. The simulation results for each improvement can be used to evaluate the pump energy loss associated with each drawback. The simulation cases are listed in Table 3.

Table-3 Simulation cases

Simulation case	Improvement A-1	Improvement A-3	Improvement B-1	Improvement B-2
	Improvement in pressure control	INV Instalation for all pumps	Adjustment of AHU Control	Adjustment of OA volume
Cost*	Low	High	Middle	Middle
CASE-0	Present simulation			
CASE-1	○	×	×	×
CASE-2	○	×	○	×
CASE-3	○	○	○	×
CASE-4	○	○	○	○

* Improvement enforcement cost

As improvement A-1 is a fundamental improvement and it is implemented in all cases. Improvement A-2 is not implemented in all cases, because BI control is not required if improvement A-1 and B-1 are successfully taken as the improvement.

Simulation was carried out using input data measured from 10 AM to 7 PM from June to September. We decided to use this data for the simulation because all of the AHUs in the department store were in full operation during this period.

Simulation Results and Effects of Improvement

Figure 16 shows the power consumption of the chilled water pump and the rate of power consumption reduction as compared to the current condition (CASE-0) for each case.

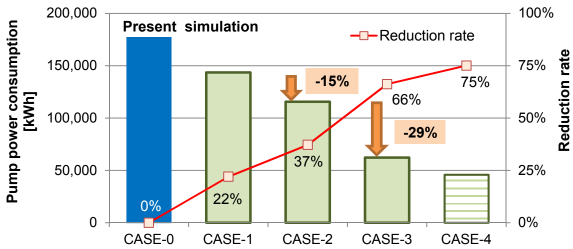


Figure-16 Secondary pump power consumption and reduction rate for each case

Improvements are carried out in priority order from lowest cost to highest cost. Reduction of the heat load (B-2) was implemented at the end.

Figure 16 shows that the reduction rate became 37% using only the low-cost improvement A-1 (22%) and B-1 (15%). The reduction rate (29%) increased when improvement A-3 was also implemented. The total reduction rate reached 75% if all improvements were implemented, i.e., CASE 4. Improvement A-3 is highly effective for multiple inverter operation, because the operation time during which more than two pumps are run was concentrated during the simulation period (June to September). Moreover, if improvements are applied to the AHU side, e.g., improvements B-1 and B-3, in addition to common improvements for pump power reduction, such as improvements A-1 and A-3, a further reduction in pump power consumption is possible. The present study is unique because comprehensive improvement effects are achieved by simulation.

CONCLUSION

Based on the results of improvement through simulation, the conditions in which the problems occur are as follows:

- Substation side:
  - -Excessive supply pressure of chilled water
- AHU side:
  - -Inappropriate set-point of the air temperature beyond the cooling coil performance
  - -Performance deterioration of the cooling coil

Simulation using the ACSES/Cx software tool revealed that the total power consumption can be reduced by 75%, even during the summertime, if improvements are carried out in order from lowest cost to highest cost to eliminate the problems described herein.

The ACSES/Cx simulation tool can simulate large air conditioning systems with high accuracy, and ACSES/Cx was demonstrated to provide a quantitative rating for the reduction effects of power consumption by improving the system. Thus, building owners and administrators of air conditioning systems are able to consider the cost-effectiveness of modifications beforehand.

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